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Comparison of Viscous and Pressure Energy Exchange in Fluid Flow Induction

by

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Submitted in partial fulfillment of the requirements for the degree of

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TABLE OF CONTENTS

LIST OF TABLES

LIST OF FIGURES

NOMENCLATURE

Primary symbols only are listed. Intermediate quantities are defined in the text.

A	-	area
D, d	-	diameter
f	-	friction factor
g	-	acceleration due to gravity
H	-	total head
h	-	head loss
K	-	constant
L	-	length
M	-	momentum
n	-	rate of contraction
p	-	pressure
Q	-	volume flow ratio
U	-	velocity
Z	-	static head

Greek Symbols

α	-	area ratio
β	-	characteristic ratio of a variable cross-section velocity flow
γ	-	specific weight
η	-	efficiency
θ	-	semicone angle

ρ - density

Subscripts

D - diffuser

jp - jet pump

mc - mixing or interaction chamber

r - ratio

p - primary

s - secondary

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I. INTRODUCTION

Jet pumps, ejectors, and eductors are members of a generic family of devices that exchange energy directly between a driving and driven fluid. No intervening mechanical system, such as a turbine or compressor, is required in the energy exchange.

The eductor uses a driving fluid that interacts or entrains a secondary fluid. The ejector uses a driving fluid to remove a secondary fluid from an enclosure. A common type, the air ejector, is used to remove air and noncondensible gases from a condenser. Another in the family, the injector, uses primary fluid to increase the head of a secondary fluid as in a feed water injector for a boiler. The water jet heat exchanger uses the primary fluid to increase or decrease the temperature of the secondary fluid as in adding heat to feed water or desuperheating steam.

Jet pumps generally consist of the following components:

1. a nozzle to introduce the high velocity primary jet,

2. a suction box or inlet section to introduce the secondary fluid,

3. a throat or fluid interaction zone where the primary and secondary flows exchange energy,

4. and a diffuser to recover the kinetic energy of the combined fluids as pressure energy.

The primary and secondary fluids of a jet pump can be either liquid or gas and a jet pump can be classified gas-gas, liquid-gas, gas-liquid, or liquid-liquid where the first term in each case is the driving fluid. Jet pumps may also be classified according to fluid phase and components. A main condenser air ejector for example, is a one phase-two component jet pump (air and steam being components), while a boiler steam-jet water injector is a two phase-one component jet pump (water and steam are two phases of the same fluid).

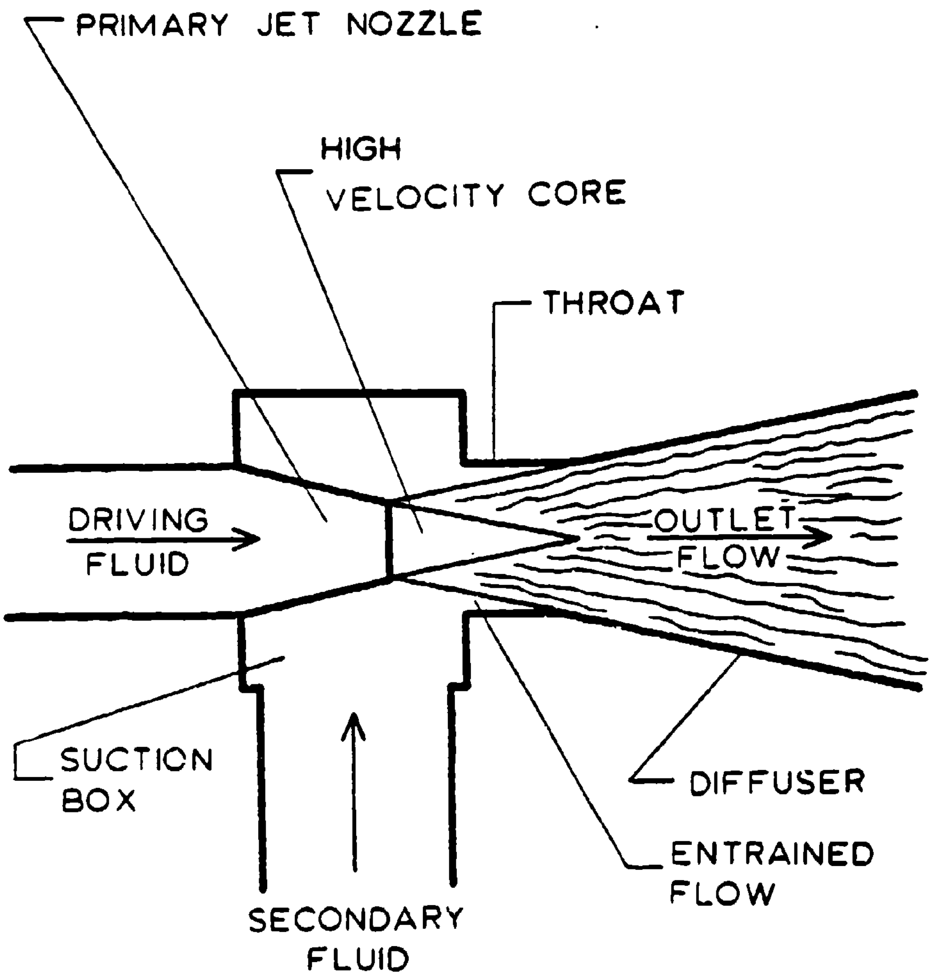

FIGURE 1: A SIMPLE JET PUMP

II. BACKGROUND

A. HISTORY

The first jet pump was used by J. Thompson in 1852 [Ref. 1]. The theory of jet pumps was advance by J. M. Ranine in 1870 [Ref. 2]. Early Jet pumps required large primary flows or pressures to produce a desired total flow. They therefore, had low efficiencies for transfering the energy of the primary fluid to the energy of the secondary fluid, hence, little interest existed for their development and use. In the 1930's Gasline and O'Brien predicted theoretical efficiencies of forty-one percent at a mass flow ratio (total mass flow to primary mass flow) of 1.2 [Ref. 3]. In the early 1940's, the United States Department of the Interior employed jet pumps with efficiencies on the order of 31 percent and flow ratios above 2.0 [Ref. 4].

In recent years jet pumps have received increased attention in applications as deep well pumping and booster pumping in the oil and energy industries, as jet pump propulsors in high performance ships, as thrust augmentors in V/STOL aircraft, and in dust collectors, exhausters, and waste gas disposal units in the environmental protection and pollution control industry.

B. THEORY OF WATER JET PUMPS

Although a water jet pump is physically a simple apparatus, the principles involved in its operation are complex and span the entire range of fluid dynamics. Some of the processes involved in the operation of a jet pump are [Ref. 5]:

1. Conversion of pressure energy to kinetic energy in the primary jet nozzle, resulting in a high velocity low pressure jet.

2. Induction of a secondary flow into the interaction zone by a pressure reduction at the primary nozzle exit.

3. Transfer of energy from the primary fluid to the secondary fluid. This occurs through an impulse of a primary fluid particle on a secondary fluid particle. On a macroscopic scale, this could be seen as entrainment of the secondary fluid through viscous transfer or energy transfer through a pressure force.

4. Conversion of kinetic energy to pressure energy of the combined primary and secondary fluids in the diffuser.

Each of the above processes is dependent on all the others. Therefore, to understand the operation of the jet pump as a whole, it is necessary to understand each of the separate processes involved and how they interact with the others.

The mixing zone of a jet pump is most often studied using applications of momentum and energy conservation at points before the streams converge and after mixing is complete

[Ref. 6]. The process taking place between these points is complex.

It is the purpose of this effort to study the interaction zone of the jet pump and to investigate the processes involved in this zone. The paper will also attempt to demonstrate what phenomena taking place in the interaction zone can be used to best advantage, to increase the total outlet flow with respect to a given primary flow input energy.

Of particular interest is the special flow interaction phenomenon known as "crypto-steady" energy exchange. While mechanical alterations of a system using crypto-steady flows are not significant, the fluid dynamics are profoundly altered. The interaction in a steady flow jet pump is caused by viscous forces, while in a crypto-steady jet pump the secondary flow acceleration is accomplished directly by pressure forces at the primary-secondary interface.

C. A SIMPLE JET PUMP

The simplest case of a jet pump will consist of a cylindrical tube drawing on a secondary incompressible fluid at atmospheric pressure at one end and discharging to the atmosphere at the other end. There is no inlet chamber and no diffuser. A primary jet nozzle located concentrically in the center of the tube will introduce the incompressible primary fluid (Figure 2), [Refs. 6, 7]. The following relationships for this simple jet pump are valid:

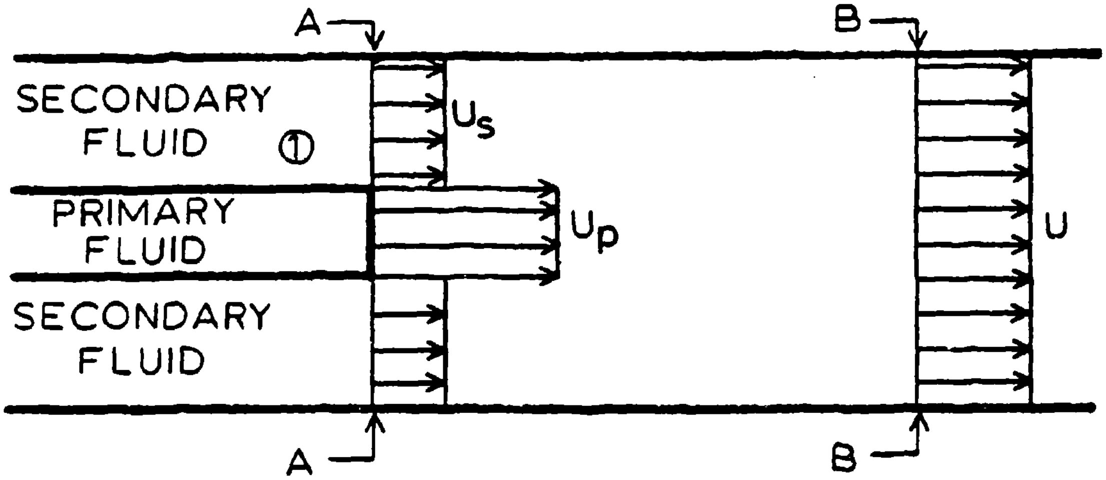

FIGURE 2: A CYLINDRICAL TUBE JET PUMP

1. Bernoulli's equation for incompressible fluids will hold from point 1 to cross section AA.

2. The continuity equation will apply for the total flow between cross section AA, where the primary jet begins interacting with secondary flow, to cross section BB, where the interaction is considered complete. The following equation pertains:

$$\int U_s dA_s + \int U_p dA_p = (A_s + A_p)U \tag{1}$$

where A_s - area of secondary flow channel at cross section AA,

U_s - velocity of secondary jet

U_p - velocity of primary jet

A_p - area of primary jet at cross section AA,

U - velocity of mixed fluids at cross section BB,

A - total [illegible] of cylinder,

$A_p + A_s =$ [illegible]

3. The momentum equation will apply to the same boundaries:

$$\int [(U_s{}^2) + (P/\rho)]dA_s + \int [(U_p{}^2 + (P/\rho)]dA_p$$
$$= (A_s + A_p)U^2 \tag{2}$$

4. For the primary fluid from primary pressure to cross section AA, (Bernoulli's equation),

$$P_p = P_{AA} + \frac{\gamma U_p{}^2}{2g} \tag{3}$$

where P_p - the total pressure of the primary fluid

P_{AA} - static pressure at cross section AA

g - acceleration due to gravity.

The outlet velocity and thus volume flow rate can be shown to be related to the primary velocity and the ratio of primary area, A, to the total cross sectional area of the interaction cylinder, [Ref. 6],

$$\alpha = \frac{A_p}{A_p + A_s} \qquad (4)$$

expresses the area ratio, the outlet velocity can be expressed by

$$U = U_p[-\alpha(1-2\alpha) + (2\alpha - 6\alpha^3 + 4\alpha^4)^{1/2}] \qquad (5)$$

Similary the outlet volume flow rate is expressed by

$$Q = \frac{Q_p}{\alpha}[-\alpha(1-2\alpha) + (2\alpha - 6\alpha^3 + 4\alpha^4)^{1/2}] \qquad (6)$$

since the primary volume flow rate is

$$Q_p = U_p A_p \qquad (7)$$

Appendix A is a complete derivation of equations (5) and (6).

D. THEORETICAL JET PUMP EFFICIENCES

As stated earlier, the jet pump works on the principle of a primary jet entraining and driving a secondary fluid. This pumping action is due to the exchange of momentum between the primary and secondary fluids. Efficiency is most often defined as useable work obtained from the system per energy input into the system. The useable work obtained from the jet pump would depend on its purpose. Where the purpose is to move a secondary fluid, the useable work obtained would be equal to the energy imparted to that fluid. Where the purpose is to produce a thrust, the useable work obtained would be equal to the energy of the total mass at the outlet of the ejector. The energy supplied to the system would always be obtained from the primary fluid. The maximum ideal pumping efficiency can be shown using the former definition:

$$\frac{\text{rate of work in moving secondary fluid}}{\text{energy supplied by primary fluid}} .$$

The momentum of the secondary fluid leaving the pump jet is

$$M = \rho A_s U_s U \qquad (8)$$

where $\rho A_s U_s = \rho Q_s$ is the mass rate of flow of the secondary fluid, and U is the discharge velocity. The rate of work in moving the secondary fluid is

$$(\rho Q_s U)U = \rho A_s U_s U^2 \qquad (9)$$

The kinetic energy supplied by the primary jet is

$$KE = \rho A_p U_p \frac{U_p^2}{2} \qquad (10)$$

where $\rho A_p U_p = \rho Q_p$ is the mass flow rate of the pirmary fluid.

The ideal pumping efficiency as per the preceeding definition is

$$\eta_{jp} = \frac{\rho A_s U_s U^2}{\rho A_s U_p \frac{U_p^2}{2}} = \frac{2Q_s U^2}{Q_p U_p^2} \qquad (11)$$

where AU = Q, the volume flow rate.

An interesting evaluation of the maximum ideal efficiency of a jet pump was proposed by Reddy and Kar [Ref. 5]. They determined that the maximum ideal efficiency would be fifty percent where volume flow rate of the primary fluid equaled the volume flow rate of the secondary fluid. Appendix B is a complete derivation of Reddy and Kar's proposal showing its limitations.

E. LOSSES

Although the ideal efficiency was shown to be 50 percent at flow rations of 1.0 and only decreasing slightly for values of Q_s/Q between 0.75 and 1.5, actual efficiencies obtained in practice are significantly lower. In 1965, a maximum efficiency of 16.1 percent for a flow ratio of 4.0 was obtained on a two stage jet pump by Hoshiet. al. [Ref. 8]. Somewhat earlier Mueller obtained an optimum efficiency for a water jet pump of 37 percent for a flow ratio of 1.5 [Ref. 9].

Some of the reduction in efficiency can be attributed to losses in individual components of the jet pump. Component losses are usually defined in terms of head loss, therefore, it will be necessary to define jet pump efficiency in terms of pumping head.

For the purpose of discussing component losses, total jet pump efficiency will be defined as:

$$\eta = \frac{Q_s(H-H_s)}{Q_p(H_p-H)} \qquad (12)$$

where Q_s - volume flow rate of the secondary fluid

Q_p - volume flow rate of the primary fluid

H_s - total head of the secondary fluid

H_p - total head of the primary fluid

H - total head of the delivered fluid.

Equation (12) is similar to that defined by equation (11) in that it compares energy added to the secondary fluid to total energy added to the system by the primary fluid. Evaluation of the fluids total head in equation (12) can be accomplished analytically by representing the total head by the sum of the fluids static head and kinetic energy. Head losses can be determined by the difference in total heads at any two points in the fluids stream.

An empirical evaluation of individual components losses may be found in Appendix C.

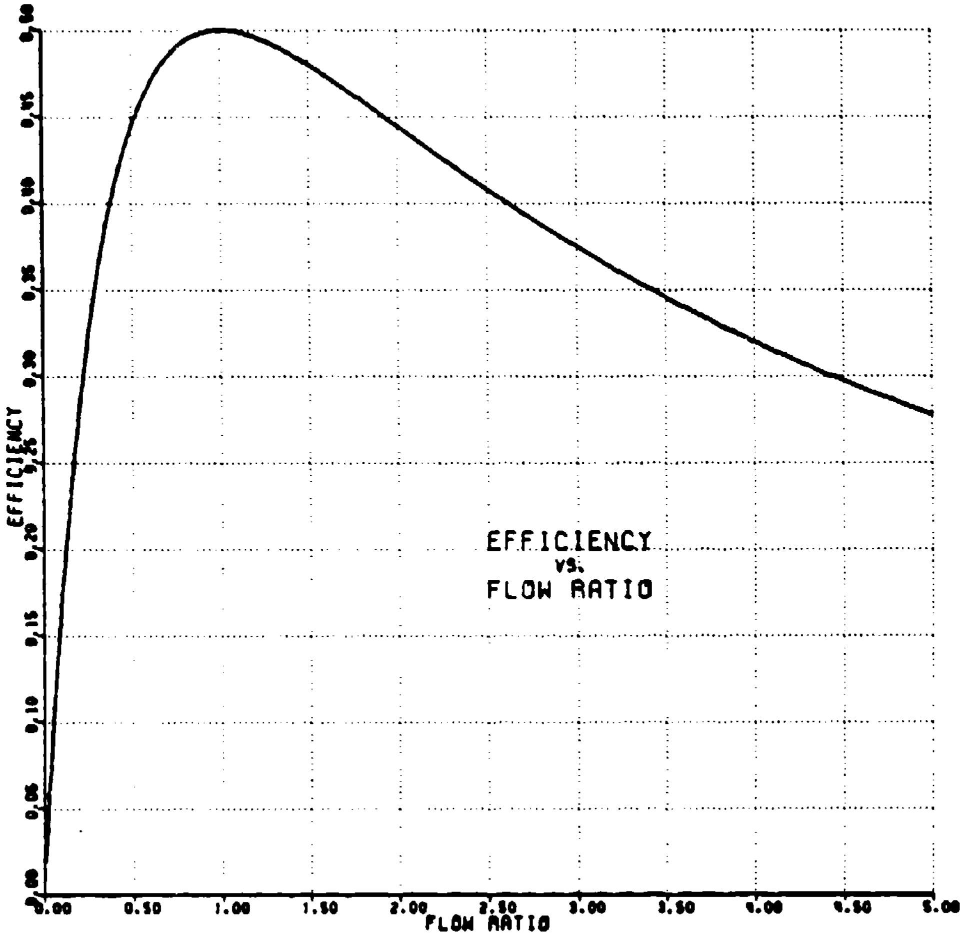

FIGURE 3: IDEAL JET PUMP EFFICIENCY VS FLOW RATIO

III. HYPOTHESIS

A. IMPROVEMENTS IN ENERGY EXCHANGE

Momentum exchange between a primary flow and a secondary flow can be accomplished by two methods; viscous entrainment and pressure exchange. Viscous entrainment is an irreversible and dissipative process. It requires a large mixing chamber, is slow and is relatively inefficient. Friction losses is the mixing chamber further contribute to total system losses, and are directly proportional to the length of the mixing chamber, as seen in equation (39). Thus, if the length of the mixing chamber can be reduced, the efficiency of the jet pump is expected to improve. Size reduction of the overall system is also attractive in certain volume and weight sensitive applications such as aircraft.

In order to reduce the size of the interaction chamber, the energy exchange rate of the primary and secondary fluids must be increased. This can be accomplished by increasing the primary-secondary interaction area and/or increasing the rate of spreading of the primary jet. Figures 4 through 8 all depict techniques used in the past to perturb the primary flow in order to increase the interaction rate.

A potentially more efficient method of primary-secondary energy exchange can be accomplished through the work of pressure forces at the primary-secondary fluid interface.

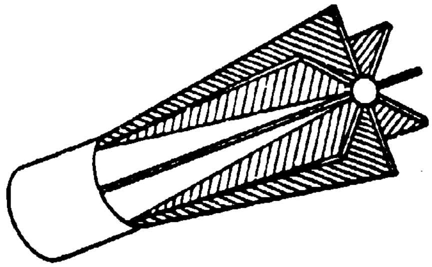

FIGURE 4

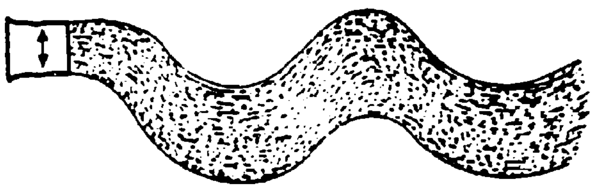

FIGURE 5

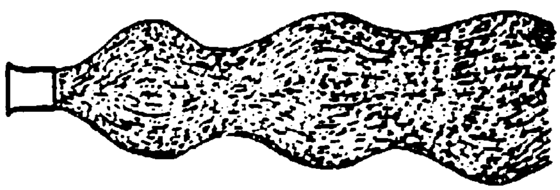

FIGURE 6

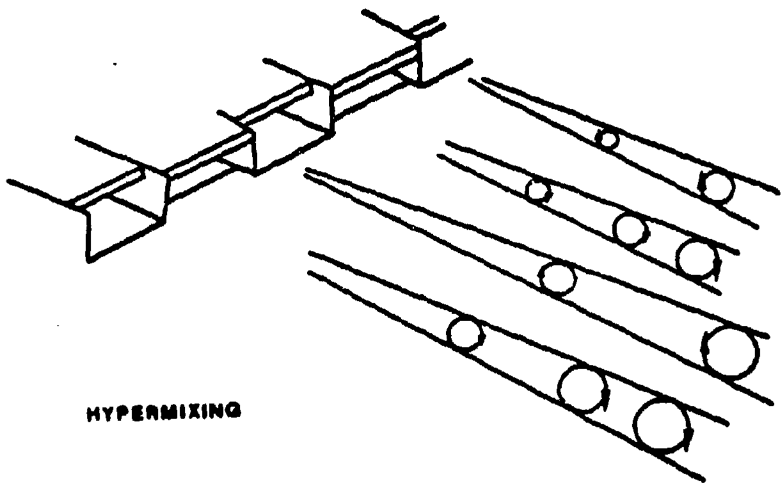

FIGURE 7: HYPERMIXING PRIMARY

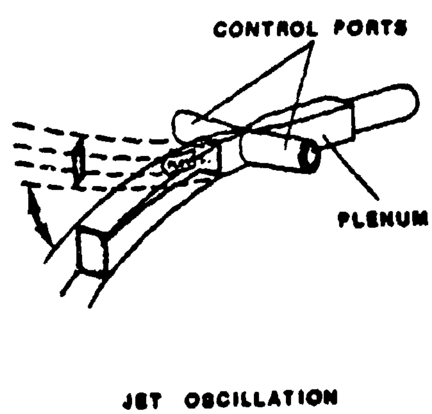

FIGURE 8: JET OSCILLATION OF PRIMARY

Essentially reversible, the work of interface pressure forces is of necessity nonsteady because no work can be done by pressure forces acting on a stationary interface. Figure 9 depicts a steady flow ejector and the pulsating or pulse tube ejector. Pulse tube ejectors have been previously investigated as thrust augmentors where the primary flow acts as a piston and energy is transferred to the secondary fluid at the pressure interface. The pulsating flow arrangement offers size as well as performance advantages as shown by Lockwood [Ref. 11].

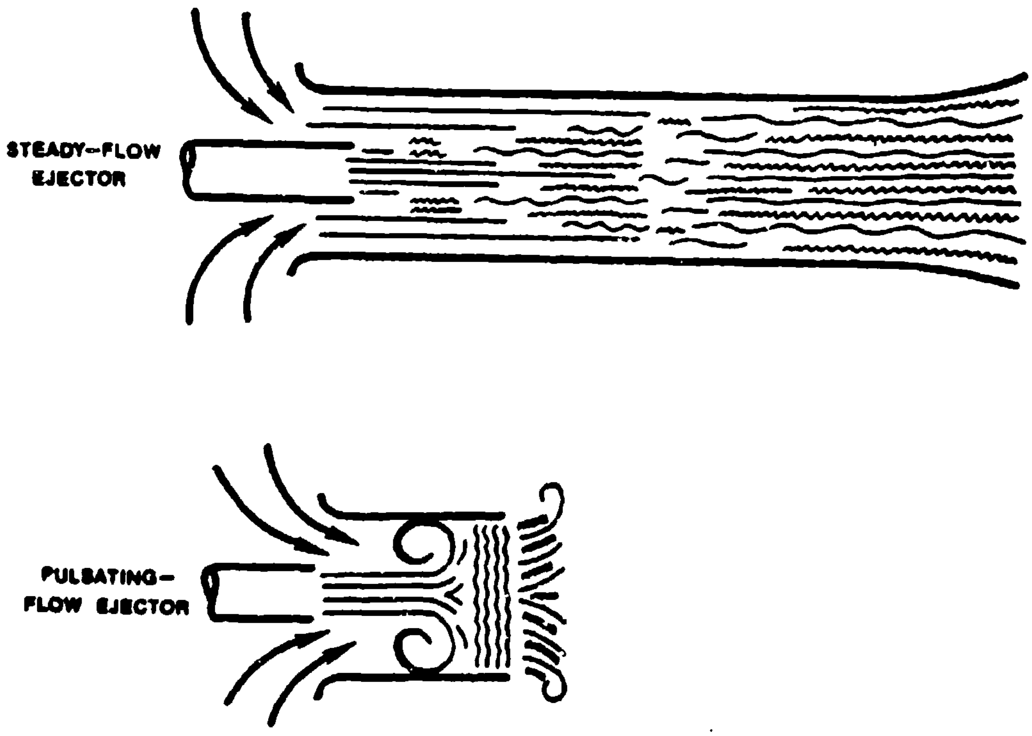

FIGURE 9: STEADY FLOW AND PULSATING FLOW EJECTORS SHOWING COMPARATIVE LENGTHS

B. CRYPTO-STEADY FLOW

In most applications, both the primary and secondary streams are steady. Thus, in order to generate a primary stream to take advantage of a nonsteady primary to secondary energy exchange, the steady primary flow wculd have to be severely perturbed. A problem is thus generated in that "chopping up" or interrupting the primary fluid introduces losses which, in all likelihood, would exceed any possible gains produced by an improved primary to secondary energy exchange. One proposed resolution of this problem is the use of a "crypto-steady" pressure exchange as desbribed by Foa [Ref. 12]:

> Crypto-steady pressure exchange is a mode of direct energy transfer between flows, based on the principle that two adjacent streams which are both isoenergetic in the same frame of reference will, in general, exchange mechanical energy in any other frame. The efficiency of this process is potentially high, because a change of frame of reference is reversible, and the associated transfer of energy is therefore nondissipative.
>
> An application of this principle to thrust or lift generation is discussed for the purpose of illustration. In this application the interacting flows are steady and isoenergetic in a rotating frame of reference but exchange energy in a stationary frame. The exchange mechanism is essentially similar to that of a turbofan, but the 'blades' are now patterns rather than bodies of abiding material.

The advantage of using crypto-steady flows in a jet pump, is that the primary flow can be generated, controlled and studied as a steady flow in the rotating frame of reference, and used to exchange energy in the stationary frame of reference in which the jet pump exists.

This can be accomplished physically by developing a primary nozzle in which the primary flow leaves with some spin or blade angle. The primary nozzle is allowed to spin freely as a result of the thrust reaction created by the primary flow (much like a lawn sprinkler). It is important to point out that although the primary flow describes a helix much like that from a propeller the motion of an individual particle of the primary flow is essentially axial.

There are several thought process that can be followed in order to explain the mechanism involved in this mode of energy exchange.

One is based on the fact that the rotating nozzle is continually moving the primary jet through the secondary flow field. Therefore, as the primary jet leaves the primary nozzle, it is constantly being introduced to low energy secondary fluid. This produces a higher velocity gradient between the primary and secondary fluids then with a non-rotating nozzle, where only the secondary fluid adjacent to the first short distance of the surface of the stationary primary jet see the full primary-secondary velocity difference. The higher velocity gradient between the primary and secondary fluids, in turn, produces a higher shear rate and thus increases the viscous interaction.

A different reasoning approach suggests that the improved energy transfer of the rotating primary nozzle system lies in another macroscopicly conceptualized phenomenon.

The rotating nozzle physically distributes the energy of the primary fluid throughout the secondary flow cross section. This process might best be described if a stream tube with infinitely thin walls is imagined in the interaction zone. Its centerline is parallel to the centerline of the rotating nozzle. A high energy pulse of primary fluid will enter the stream tube each time a nozzle passes its entrance. This primary fluid element will have slower moving secondary flow immediately ahead of it. The primary and secondary fluid elements will then exchange energy across their interface; the primary slowing while the secondary accelerates. Behind the primary element secondary fluid is drawn into the pseudo-stream tube. As the next primary nozzles crosses the stream tube another primary pulse will enter. Energy will be exchanged between the primary and secondary fluids at their interface much as energy is exchanged between the primary and secondary fluids in a pulse jet. This energy exchange through a pressure force is essentially reversible and thus should increase the efficiency of the system.

Undoubtedly, neither of the above descriptions is complete in explaining the phenomenon involved. In all likelihood, there appears to be only a subtle difference between a relatively random particle momentum exchange in the viscous interaction concept and a more ordered energy exchange in the crypto-steady or pseudo-blade system.

An experiment was developed to study the interaction zone of the jet pump. The primary purpose of the experiment was to compare the effects of rotating and nonrotating primary jet streams on the fluid interactions. In addition, the effects of changing primary energy and nozzle rotational speed are also investigated.

C. THE EXPERIMENTAL APPARATUS

To study the effect of improved energy transfer in the jet pump, many of the components of the pump not directly related to mixing, including the secondary inlet nozzle and outlet diffuser, have been eliminated. The experimental apparatus consists of a primary tube to inject the primary flow, various interchangable nozzles to explore steady and crypto-steady primary flow effects, and the interaction chamber. The secondary nozzle and outlet diffuser have been eliminated as not pertinent to this study. The dimensions of the mixing chamber, secondary suction box, and flow outlet were kept constant to eliminate unnecessary variables. The primary tube was extended throughout the mixing chamber to keep the cross section of the mixing chamber constant and to improve the observation of the phenomenon taking place.

The secondary flow suction and total flow discharge were maintained in the same horizontal plane and discharged water into the same tank to ensure constant and equal static heads at those points. The total mass flow was measured by an elbow flow meter just prior to discharge.

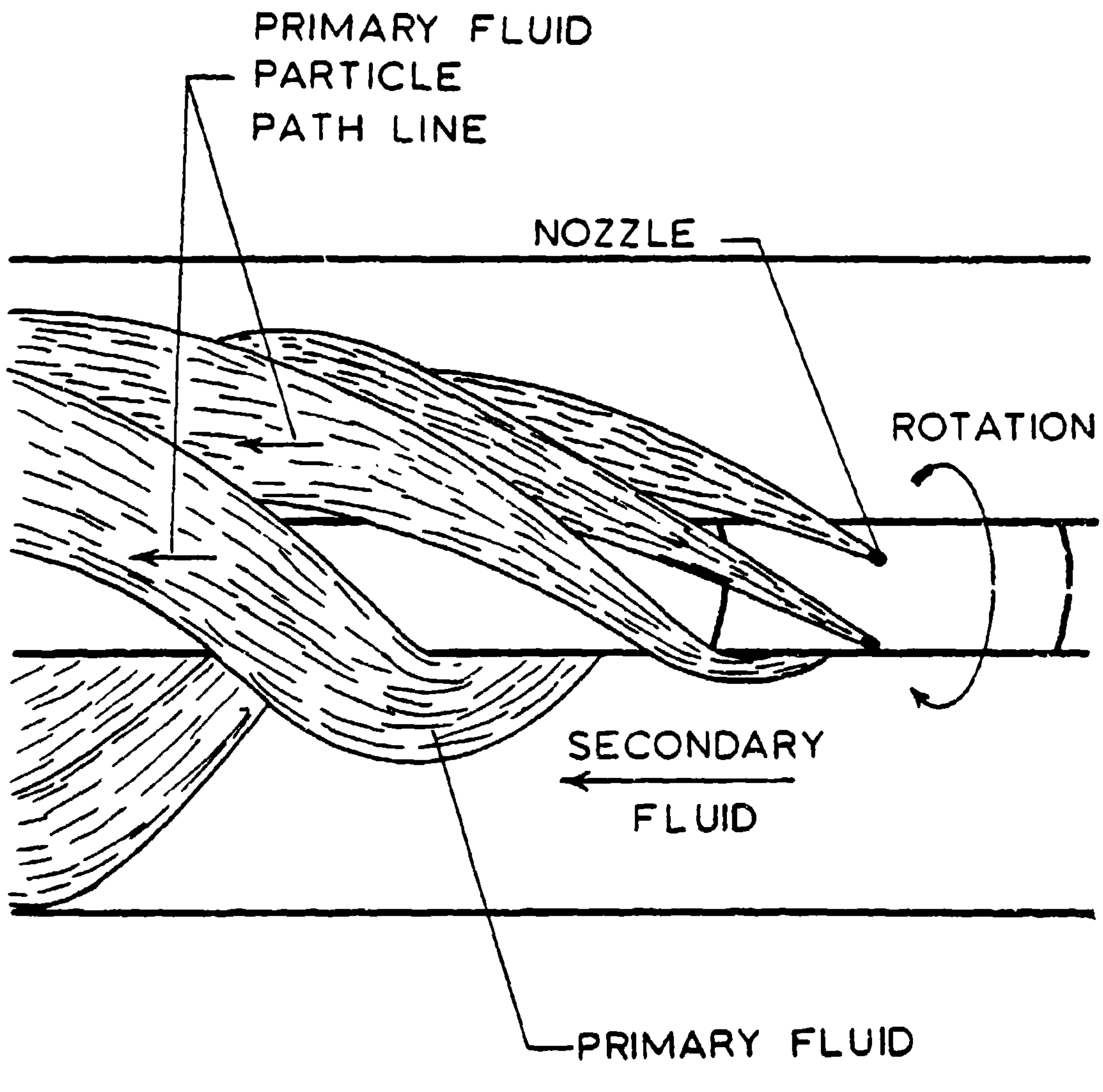

FIGURE 10: DIAGRAM OF INTERACTION ZONE IN A CRYPTO-STEADY JET PUMP

The primary flow was introduced about midway through the mixing chamber. Mass flow was measured by a strain gage flow meter and pressure was measured by both a bourdon tube pressure gage and a strain gage pressure transducer. Figure 11 is a schematic of the experimental apparatus. Figure 12, is a photograph of the actual experimental system.

Three primary nozzles were employed in the experiement. A coning angle of five degrees was used on all primary nozzles. The coning angle is that angle at which the primary fluid is injected into the secondary with respect to the axis of secondary flow. In cylindrical coordinates the coning angle is represented by ϕ, the angle measured down from the centerline of the secondary flow (x-axis) to the line representing the primary flow.

Blade angles of zero, twenty, and thirty-five degrees were used on the different primary nozzles. The blade angle is also measured with respect to the centerline of the secondary flow (x-axis), however, it is offset from that axis by the radius of the primary nozzle and is perpendicular to that radius. It is the blade angle that causes the primary flow to induce torque to the primary nozzle. Figure 13 describes the blade angle.

The primary fluid enters the interaction zone through four 1/8 inch diameter holes in the primary nozzles. Secondary flow suction and total flow discharge were each accomplished through two inch inside diameter PVC pipe. The interaction chamber

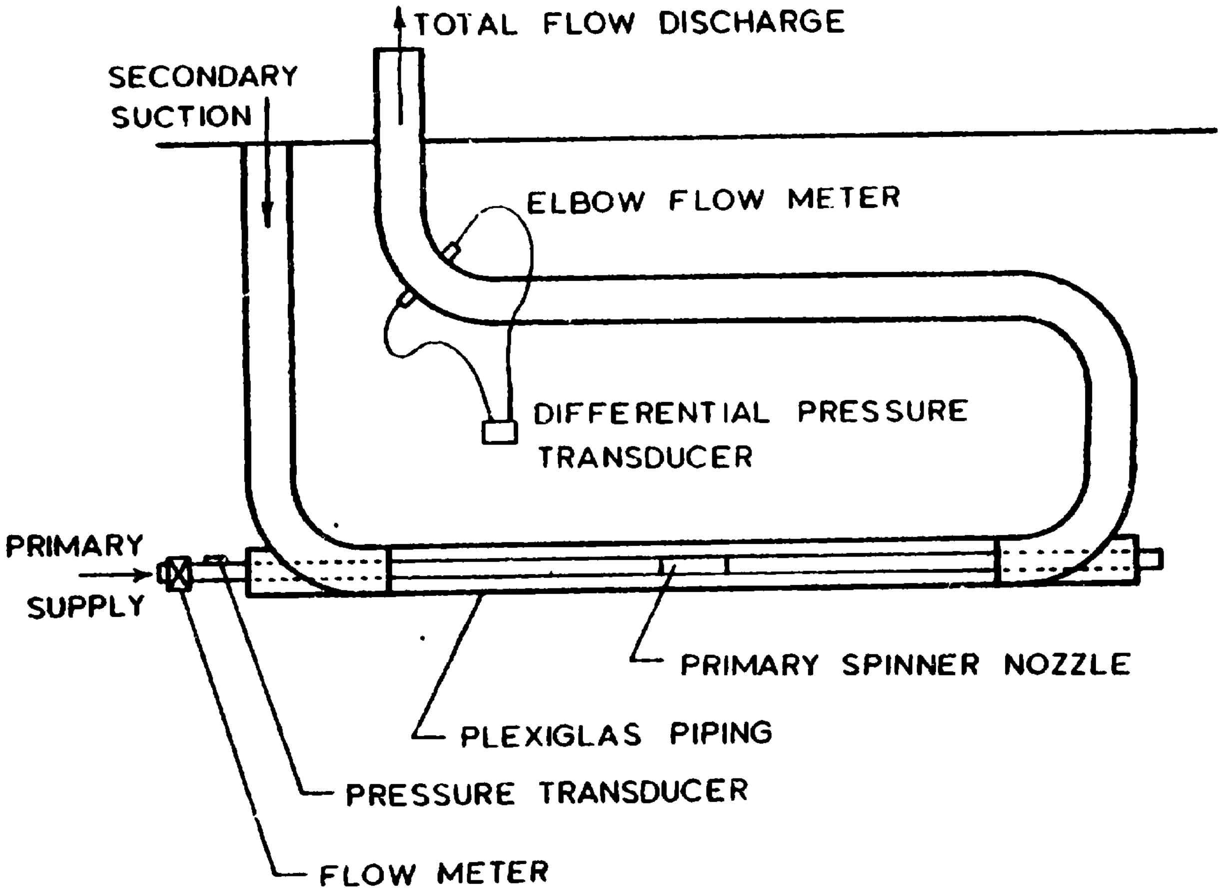

FIGURE 11: SCHEMATIC OF EXPERIMENTAL APPARATUS

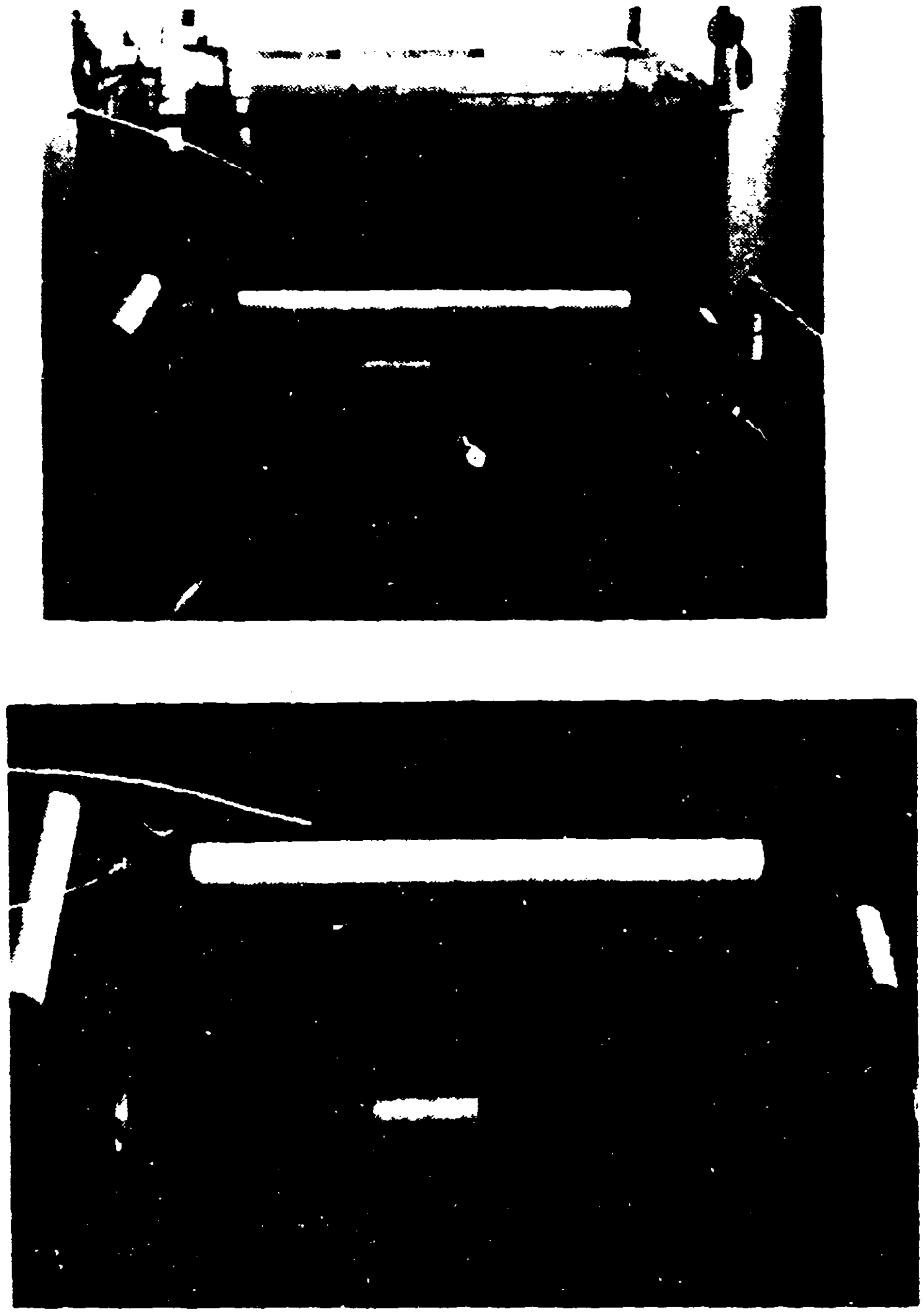

FIGURE 12: THE EXPERIMENTAL APPARATUS

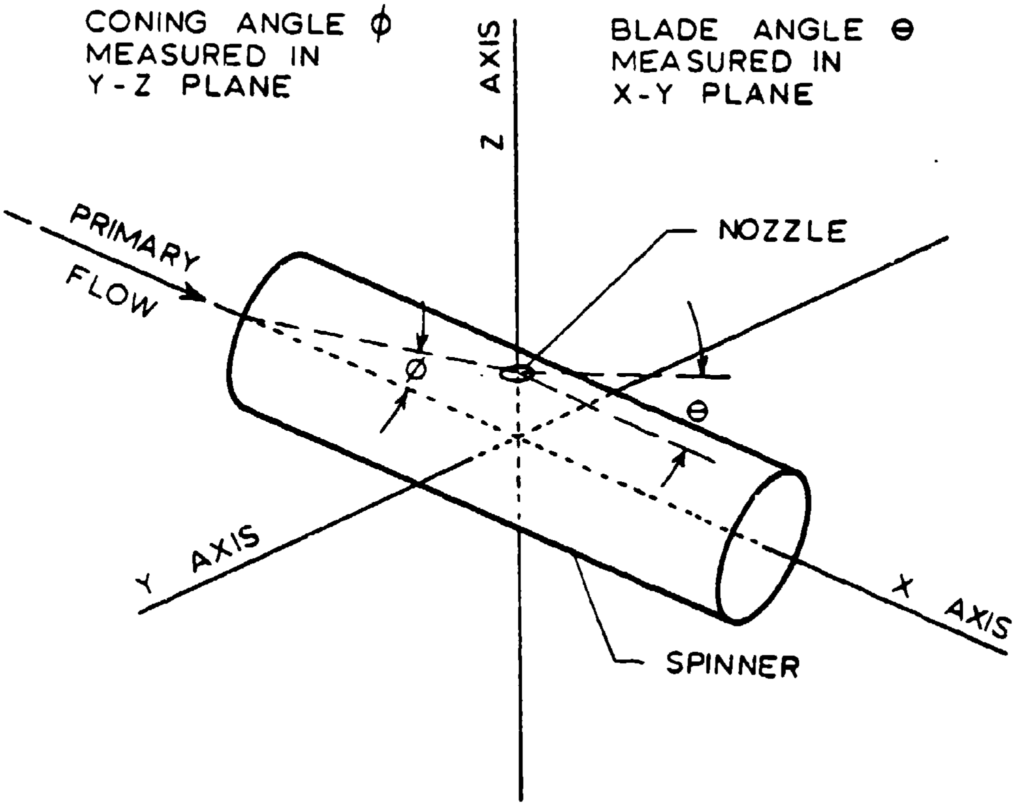

FIGURE 13: SCHEMATIC OF ROTATING PRIMARY NOZZLE

was manufactured from two inch inside diameter plexiglass pipe and the primary supply pipe extending through the interaction chamber was of one inch outside diamter aluminum pipe. A constant head in the supply/discharge tank was maintained at ten inches of water from the centerline of the secondary suction/ total discharge lines by means of a stand-pipe in the constant head tank.

The secondary flow suction box was located immediately upstream of the primary nozzle. It was twenty inches in length and of the same cross-sectional dimensions as the interaction chamber to provide for a reasonably steady secondary flow prior to interaction. The interaction chamber was also twenty inches long to allow for a complete primary to secondary energy transfer. Twenty-eight inches of PVC pipe was installed prior to the elbow flow meter to allow for a steady flow through that device.

A dye injection system was incorporated into the primary supply line to identify the primary and secondary fluids in the interaction chamber.

FIGURE 14: A 20° DEGREE BLADE ANGLE ROTATING PRIMARY NOZZLE. PRIMARY FLUID JET IS SHADED WITH DYE

IV. OBSERVATIONS

A. EXPERIMENTAL RESULTS

Several experimental runs were made with each of the primary nozzles. Primary mass flow, primary pressure and total mass flow, were measured for each of the nozzles at various primary supply pressures.

The ratio of total mass flow to primary mass flow is used as an indicator of performance of each of the three nozzles for comparison purposes. The mass flow ratio was plotted as a function of primary pressure for each of the nozzles, (Figure 15). There is almost no effect of primary pressure on the mass flow ratio of the zero degree blade angle nozzle. The slight drop as primary pressure incrases can be attributed to increased nozzle losses at higher flow velocities at the higher primary pressures. For both the twenty and thirty-five degree blade angle the mass flow ratio increased dramatically with increased primary pressure. In addition, the mass flow ratio increases with increasing blade angle when the same primary pressure was applied to each nozzle.

The mass flow ratios for the zero degree blade angle nozzle, operating only on a viscous interaction basis between the primary and secondary fluids, ranged from 3.88 to 3.59, decreasing as primary fluid pressure (and thus velocity) increased. The mass flow ratios of the twenty degree blade angle primary nozzle ranged from 4.06 to 4.70 increasing as primary pressure (and thus nozzle rotational speed) increased. The mass flow ratios for the thirty-five degree blade angle spinner ranged from 4.30 to 5.44 again increasing as primary pressure increased.

The interaction efficiency of each configuration of the experimental jet pump is calculated using equation (23). Interaction efficiency versus blade angle is plotted for various primary pressures in Figure 16.

The improvements in efficiency are also dramatic; from 1.62 to 2.12 percent for various pressures in the zero degree blade angle nozzle, 2.43 to 3.98 percent for the twenty degree blade angle nozzle, and 2.98 to 6.42 percent for the thirty-five degree angle nozzle. Although these numbers may appear small, the relative improvement in efficiency, especially at the higher primary pressures is impressive. By introducing a pressure force energy exchange between the primary and secondary fluids, the improvement in interaction efficiency was more than tripled at primary pressures over twenty pounds per square inch.

It appears that the increased flow ratio at constant primary pressure and with increasing nozzle blade angles is caused by the introduction of a pressure force interaction between the primary and secondary fluids. The basis for this assumption is the increased efficiency observed as the blade angle increases. Since the zero degree nozzle has no spin, the primary fluid merges with the secondary fluid in parallel

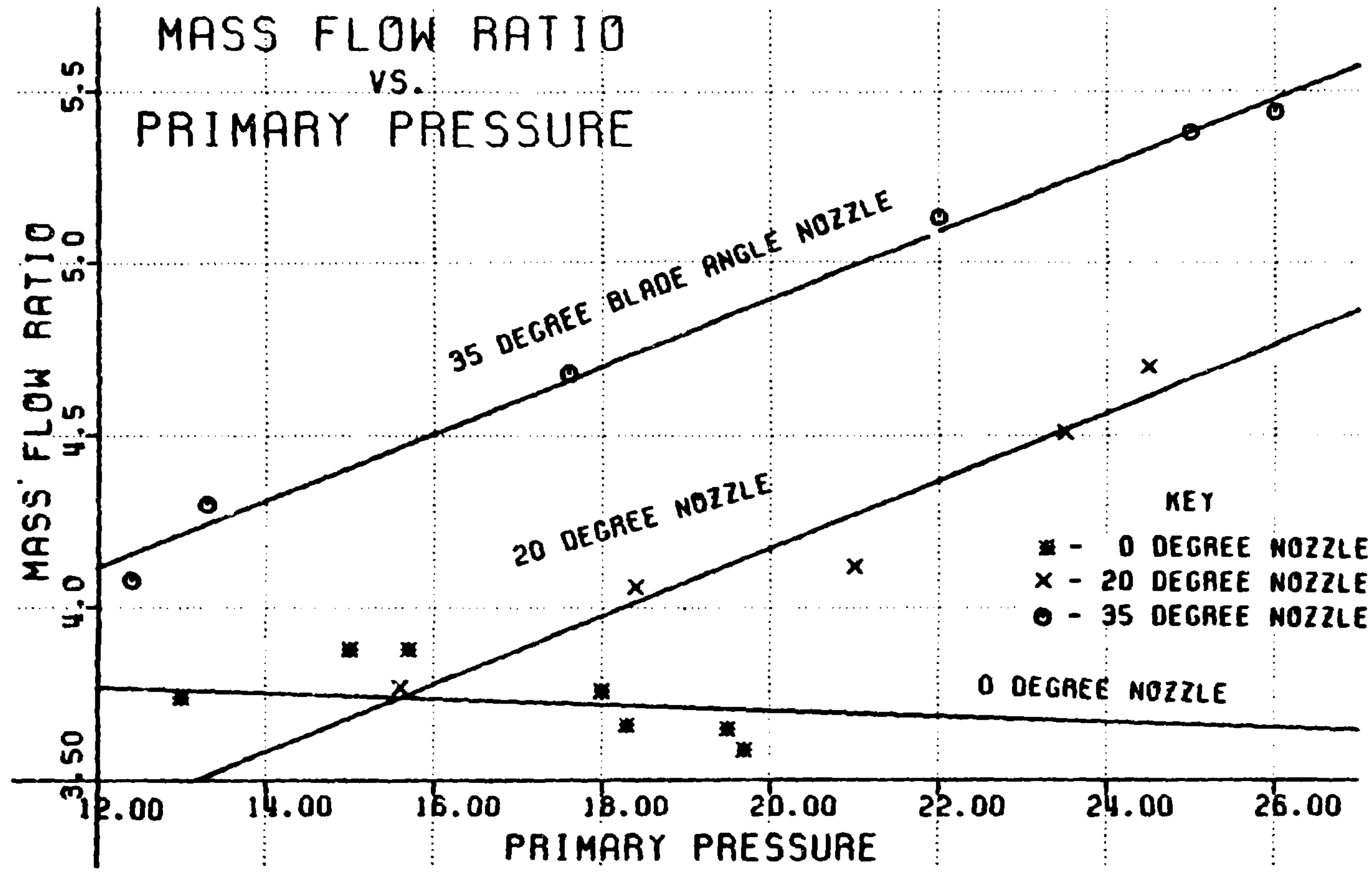

FIGURE 15: MASS FLOW RATIO ($\dot{m}/\dot{m}_p$) VS PRIMARY PRESSURE FOR VARIOUS BLADE ANGLE NOZZLES IN THE EXPERIMENTAL APPARATUS

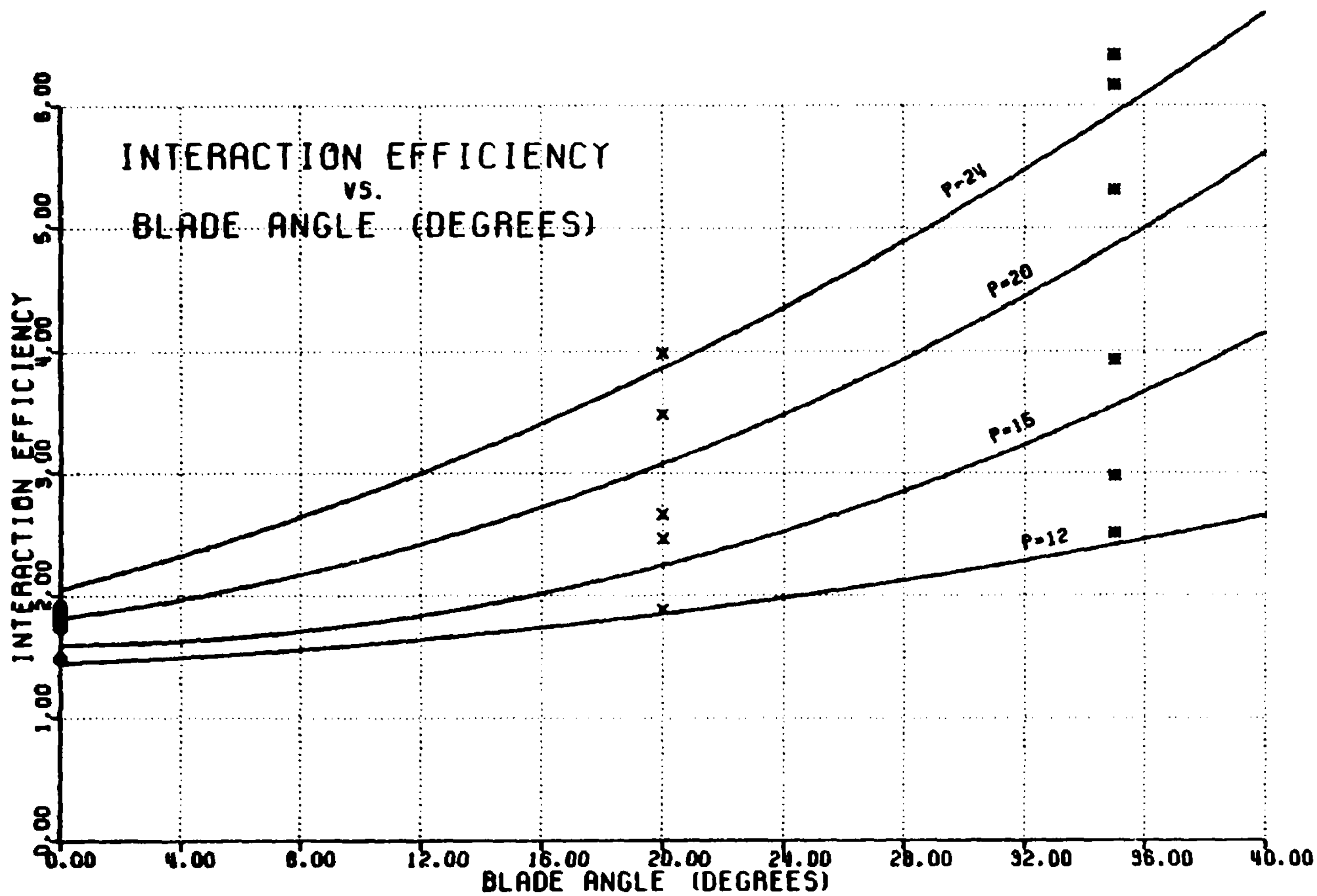

FIGURE 16: INTERACTION EFFICIENCY VS BLADE ANGLE FOR VARIOUS PRIMARY PRESSURES IN THE EXPERIMENTAL APPARATUS

and continuous streams. All primary particles follow the same stream line. The method of energy exchange between the primary and secondary fluids is by viscous entrainment only. Spin is imparted to the twenty and thirty-five degree nozzles by the reaction as a result of the tangential momentum component of the primary flow. Although the individual particles of fluid have pathlines parallel to the secondary axis, each consecutive primary particle from one revolution of the primary nozzle has its own streamline. The streamlines are repeated every revolution of the primary nozzle, but the particles are delayed by the period of the nozzle. By isolating one streamline, one would see a series of primary pulses separated by secondary flow. If the operation of the jet pump could be frozen in time, the flow field would have the appearance illustrated in Figure 10. The primary fluid would form a helix with all particles of the primary fluid moving parallel to the secondary axis. Secondary fluid is trapped within this helix and is "pushed" along by a pressure exchange. It is this introduction of an energy exchange process through the action of pressure forces that has produced the increase in mass flow ratio. As the helix gets tighter, as from the twenty to thirty-five degree nozzles, the angular speed of the nozzle increases and the number of pressure impulses from the primary increase. It is expected, however, that there is an optimum blade angle for in the limit where the blade angle approaches ninety degrees the primary motion would be purely radial, therefore, there could be no axial component of the primary jet and, hence no energy exchange in the axial direction.

Attempts were made to visualize the primary-secondary interface in the interaction zone by dying the primary fluid. In this way the primary fluid could be distinguished from the secondary fluid after it entered the interaction chamber. By careful observation the primary fluid could be identified for approximately twelve primary nozzle diameters down the interaction chamber from the non-rotating primary nozzle, although it was spreading rapidly. In either of the rotating primary nozzles the colored primary fluid quickly disappeared in a cloud of dye shortly (two to three primary nozzle diameters) after leaving the primary nozzle, indicating an increased rate of interaction.

Appendix D is a list of raw data.

B. LOSSES IN THE ROTATING PRIMARY NOZZLE

Energy losses in the spinning nozzle can have serious detrimental effects on the operation of a crypto-steady jet pump. If friction inhibits the rotation of the primary nozzle, the flow of the primary fluid after it leaves the nozzle will not be parallel to the secondary flow axis. A 'frictionless' speed of rotation was calculated by solving for the component of the mass rate of flow perpendicular to the nozzle axis and assuming all bearings and surfaces were frictionless. If the primary fluids velocity were truly axial the tangential speed of the nozzle should be equal and opposite to the component of flow perpendicular to the nozzle axis. As the rotation of the nozzle slow from its calculated frictionless speed to no rotation, the primary to secondary energy exchange difference diminishes to zero. It was observed that for the maximum primary flow of the thirty-five degree nozzle (6.1 GPM), the frictionless speed of rotation would be 5413 RPM. The spin of the primary nozzle was measured at 605 RMP. A frictionless twenty degree nozzle would spin at 3330 RPM for a primary flow of 6.5 GPM. In the actual case, the speed of the primary nozzle was measured at 250 RPM.

As friction slow the primary nozzle down, the individual particles of primary flow enters the secondary at an angle to the flow. The streamlines formed by the primary takes the shape of a helix and the distance between primary impulses increases. It will be, therefore, necessary to devote thought to reducing friction in the spinning primary nozzle to as small a value as possible.

Another major energy loss from the primary fluid occurs during the transition from the stationary primary supply line to the rotating nozzle. Whereas the primary nozzle in a stationary jet pump can be designed to make losses almost negligable, the losses in the rotating nozzle can be high.

V. CONCLUSIONS AND RECOMMENDATIONS

The mass flow ratio nad the efficiency of a jet pump can be substantially increased by introducing a rotating flow. It appears that the rotating primary causes an energy exchange from the primary fluid to the secondary fluid through an interface pressure force. Non-rotating jet pumps transfer energy through viscous interaction. The reversible nature of work accomplished through a pressure exchange is inherently more efficient then the nonreversible work accomplished through viscous interaction.

It was experimentally demonstrated that the ratio of total discharged mass flow to primary inlet mass flow can be greatly increased through the use of a rotating primary nozzle. The efficiency of the fluid interaction with a rotating primary nozzle was increased over the efficiency of the viscous fluid interaction.

It is expected that the efficiency of fluid interaction in a rotating primary jet pump can be further improved if:

1. design of the rotating nozzle is improved to reduce losses in the primary nozzle to a minimum;

2. establish an optimum area ratio, for best interaction efficiency;

3. establish an optimum blade and coning angle for maximum interaction efficiency; and,

4. introduce a secondary nozzle to establish an optimum secondary to primary mass flow ratio for maximum efficiency.

Finally, future work will require investigation of the intricacies of the interaction zone. This could be accomplished by inserting a ram pressure probe, hot wire anomometer or a laser doppler anonometer into the primary stream at various points within the iteraction zone to determine how and where the primary fluid interacts with the secondary fluid.

APPENDIX A

DERIVATION OF VELOCITY AND VOLUME FLOW RATIO EQUATIONS

The following derivation for Equations (5) and (6) is taken from Von Karman [Ref. 6]. In these equation U denotes the axial component of the velocity of the primary flow. The secondary flow well upstream of cross section AA is at rest and at atmospheric pressure. All pressures are relative to atmospheric pressure. Finally, U, the outlet velocity is assumed to be distributed over cross section BB. Since the velocity distribution is uniform Equations (1) and (2) can be written:

$$A_s U_s + A_p U_p = (A_s + A_p) U \qquad (13)$$

$$A_s (U_s^2 + P_{AA}/\rho) + A_p (U_p^2 + P_{AA}/\rho) = (A_p + A_s) U^2 \qquad (14)$$

Since $\frac{P_{AA} g}{\gamma} = \frac{P_{AA}}{\rho} = -\frac{U_s^2}{2}$ according to Bernoulli's equation, Equation (14) becomes:

$$A_s \frac{U_s^2}{2} + A_p U_p^2 - A_p \frac{U_s^2}{2} = (A_p + A_s) U^2 \qquad (15a)$$

$$\text{or} \quad (A_s - A_p) \frac{U^2}{2} + A_p U_p^2 = (A_p + A_s) U^2 \qquad (15b)$$

Solving Equation (13) for U_s and substituting into Equation (15b) gives

$$\left(\frac{A_s - A_p}{2}\right)\left(\frac{(A_s - A_p)U - A_pU_p}{A_s}\right)^2 + A_pU_p^{\,2} = (A_p + A_s)U^2 \qquad (16)$$

Equation (16) can then be solved quadratically for the outlet velocity as

$$U = U_p[-\alpha(1-2\alpha) + (2\alpha - 6\alpha^3 + 4\alpha^4)^{1/2}] \qquad (5)$$

where α is the ratio of the primary jet area to the total flow area,

$$\alpha = \frac{A_p}{A_p + A_s} \qquad (4)$$

In many pump jet applications the primary jet area is small in comparison to the total flow area, thus

$$U = U_p\ [\ -\alpha + \sqrt{2\alpha}\ ] \qquad (17)$$

is an adequate approximation of the outlet velocity for small area ratios.

The discharge volume flow rate, Q, can be found where

$$Q_p = U_pA_p \qquad (18)$$

and

$$Q = U(A_s + A_p) \qquad (19)$$

Substituting into Equation (5) gives

$$Q = \frac{Q_p}{\alpha}\ [-\alpha(1-2\alpha) + (2\alpha-6\alpha^3 + 4\alpha^4)^{1/2}] \qquad (6)$$

as a result. By eliminating higher order terms for small area ratios

$$Q = \frac{Q_p}{\alpha}[- \alpha + \sqrt{2\alpha}] \tag{20}$$

will provide satisfactory results.

Figures 17 and 18 are plots of the discharge velocity/primary jet velocity ratio and discharge flow rate/primary mass flow rate ratio versus the jet area/total area ratio. As can be seen from Figure 17, the approximate solution can only be justified for values of α less than 0.2. In jet pump applications where α is greater than 0.2, the exact solution, although algebraicly involved, will have to be employed.

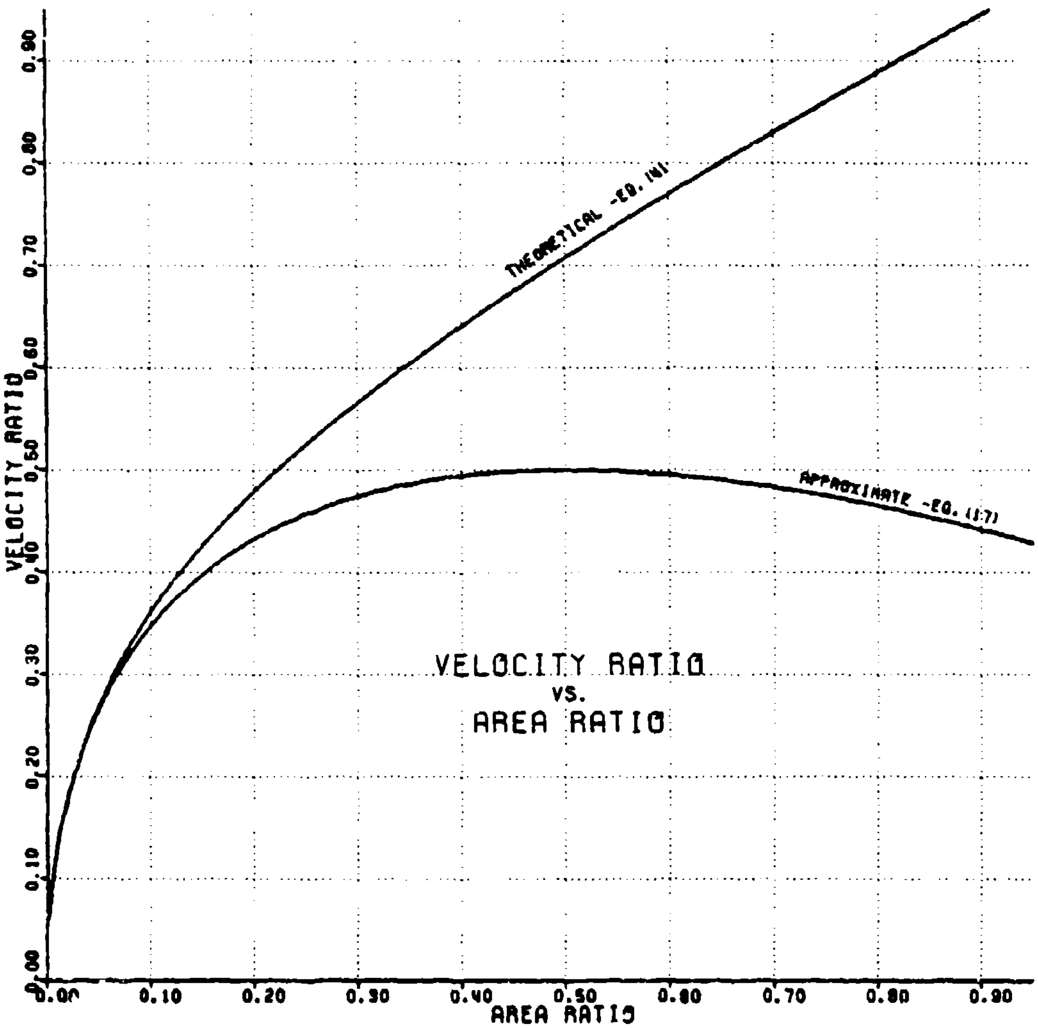

FIGURE 17: VELOCITY RATIO (U/U_p) VS AREA RATIO (A_p/A) FOR CYLINDRICAL TUBE JET PUMP

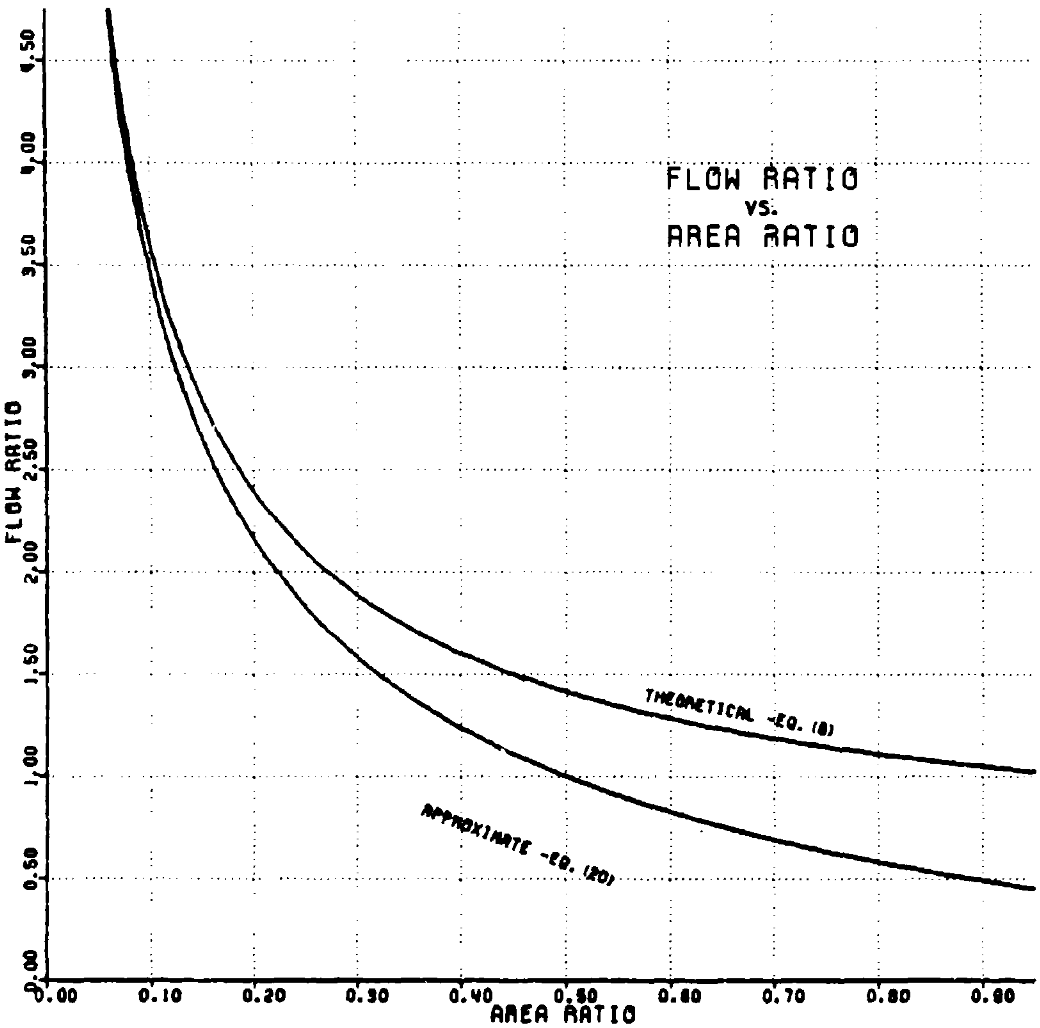

FIGURE 18: VOLUME FLOW RATIO (Q/Q_p) VS AREA RATIO (A_p/A) FOR CYLINDRICAL TUBE JET PUMP

APPENDIX B

DERIVATION OF MAXIMUM IDEAL JET PUMP EFFICIENCY

An interesting evaluation of the maximum efficiency of jet pump was proposed by Reddy and Kar [Ref. 5] using the continuity and momentum equations. Of the several assumptions he makes including constant and equal fluid densities and negligible losses in the throat/mising section of the jet pump, by far the weakest is neglecting the secondary fluid's momentum.

The evaluation follows. With the assumption of a negligible secondary momentum, the momentum equation through the jet pump is:

$$\rho Q_p U_p = \rho QU \tag{20}$$

substituting the continuity equation,

$$\rho Q = \rho Q_p + \rho Q_s \tag{21}$$

into Equation (24) and factoring out the density produces

$$Q_p U_p = (Q_s + Q_p)U \tag{22}$$

solving for the flow ratio

$$\frac{Q_s}{Q_p} = \frac{U_p - U}{U} \tag{23}$$

substituting Equation (23) into Equation (11)

$$\eta_{jp} = \frac{2(U_p - U)U}{U_p^2} \tag{24}$$

The jet pump efficiency will be maximum when $\frac{d\eta_{jp}}{dU} = 0$:

$$\frac{d\eta_{jp}}{dU} = \frac{2U_p - 4U}{U_p^2} = 0 \tag{25}$$

or

$$U_p = 2U \tag{26}$$

The maximum efficiency will therefore be

$$\eta_{jp} = \frac{2(2U-U)U}{(2U)^2} = \frac{2U^2}{4U^2} = .5 \tag{27}$$

and the flow ratio

$$\frac{Q_s}{Q_p} = \frac{2U-U}{U} = 1 \tag{28}$$

The ideal interaction efficiency as a function of flow ratio is plotted in Figure 3.

When using the above derivation care must be taken to ensure the assumption of negligible secondary momentum is valid. Using the continuity equation and solving for Q_s/Q_p in terms of area ratio, α, and velocity ratio, U/U_p, produces

$$\frac{Q_s}{Q_p} = \frac{Q-Q_p}{Q_p} = \frac{Q}{Q_p} - 1 = \frac{AU}{A_pU_p} - 1 = \frac{1}{\alpha}\left(\frac{U}{U_p}\right) - 1 \tag{29}$$

when from Equation (27), $U/U_p = 0.5$

$$\frac{Q_s}{Q_p} = \frac{1-2\alpha}{2\alpha} \tag{30}$$

The jet pump efficiency then is

$$\eta_{jp} = 2\left(\frac{Q_s}{Q_p}\right)\left(\frac{U}{U_p}\right)^2 = \frac{1-2\alpha}{4\alpha} \qquad (31)$$

For the relationship to have reasonable meaning α must lie between 1/6 ($\eta_{jp} = 1$) and 1/2 ($\eta_{jp} = 0$). In addition the secondary to primary velocity ratios can be found using the equation

$$\frac{U_s}{U_p} = \frac{Q_s A_p}{Q_p A_s} = \frac{Q_s}{Q_p}\left(\frac{\alpha}{1-\alpha}\right) \qquad (32)$$

If, according to Equation (28), Q/Q equals one, the area ratio, α, must be small to ensure the assumption that the secondary flow momentum is correct.

APPENDIX C

COMPONENT LOSSES

The following derivation of jet pump was adapted from Reddy [Ref. 13] and Reddy and Kar [Ref. 5]. Loss equations for the primary nozzle, secondary nozzle and outlet diffuser were emperically developed by Reddy [Ref. 13].

A. PRIMARY LOSSES

The primary flow line consists of a straight pipe leading from a pressure source to a primary outlet nozzle. The Darcy-Weisback equation is used to express friction losses in this line.

$$h_l = f \frac{L_1}{D_1} \frac{U_1^2}{2g} \qquad (33)$$

where h_1 -head loss

f -friction factor

L_1 -length of pipe

D_1 -inside diameter of pipe

U_1 -primary flow velocity in pipe

g -acceleration due to gravity.

Using the continuity equation to express Equation (33) in terms of the outlet primary velocity

$$h_1 = f \frac{L_1}{D_1}\left(\frac{A_p}{A_1}\right)^2 \frac{U_p^2}{2g} \qquad (34)$$

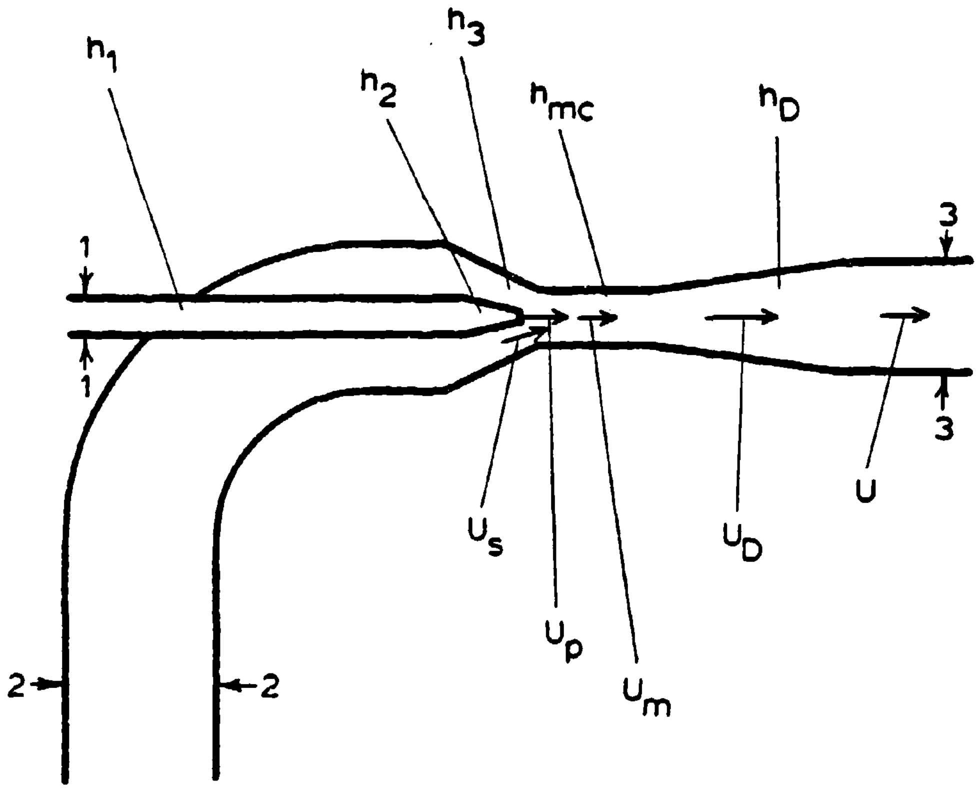

FIGURE 19: JET PUMP INDICATING LOSSES

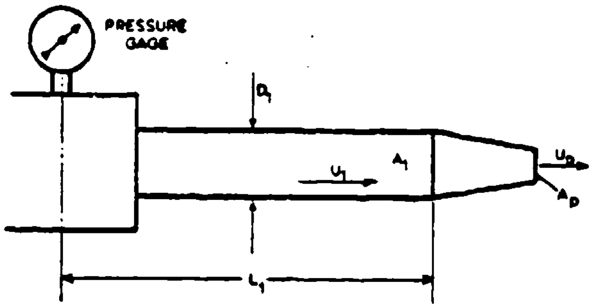

FIGURE 20: PRIMARY LINE

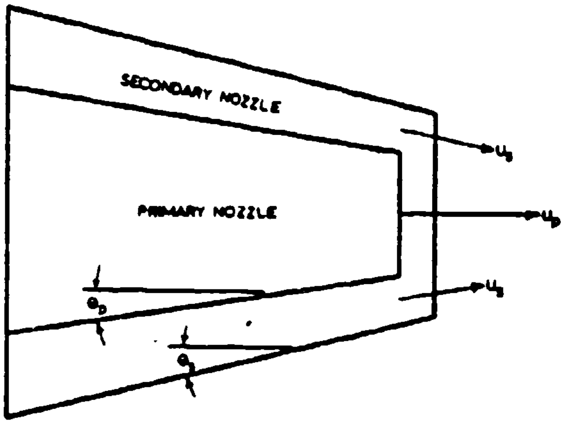

FIGURE 21: PRIMARY AND SECONDARY NOZZLES

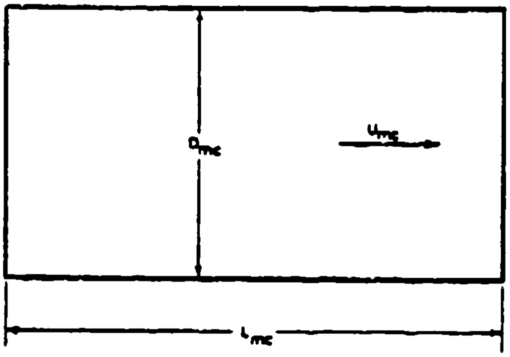

FIGURE 22: MIXING CHAMBER

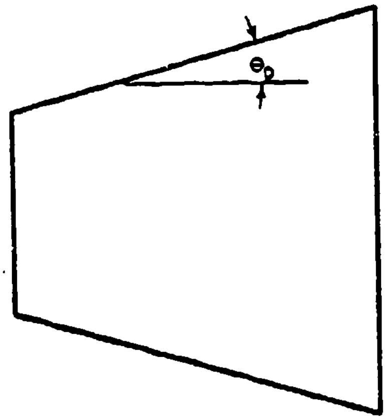

FIGURE 23: DIFFUSER

The head loss in the driving nozzle is expressed by

$$h_2 = \frac{f}{8} \cot \theta_p \left(1 - \frac{1}{n_p^2}\right) \frac{U_p^2}{2g} \qquad (35)$$

where n_p - rate of contraction of area in primary

θ_p - semicone angle of the primary nozzle

B. SECONDARY FLOW LOSSES

An equation similar to Equation (35) is used to express the head loss through the secondary nozzle, the flow converges annularly and the expression for the head losses through the secondary nozzle becomes:

$$h_3 = \left[\frac{f}{8} \cot \theta_p \left(1 - \frac{1}{n_p^2}\right) + \frac{f}{8} \cot \theta_s \left(1 - \frac{1}{n_s^2}\right)\right] \frac{U_s^2}{2g} \qquad (36)$$

where n_s - rate of contraction of area in the secondary nozzle

θ_s - semicone angle of the secondary nozzle.

If the velocity ratio is defined as

$$U_r = \frac{U_s}{U_p} \qquad (37)$$

Equation (36) can be defined in terms of the primary velocity

$$h_3 \left\{\frac{f}{8} U_r^2 \left[\cot \theta_p \left(1 - \frac{1}{n_p^2}\right) + \cot \theta_s \left(1 - \frac{1}{n_s^2}\right)\right]\right\} \frac{U_p^2}{2g} \qquad (38)$$

Bendes in the primary and/or secondary lines, penetrations for sensors and other flow disruptions will also cause lossess for which must be accounted. Theoretical expressions for these losses can be found in Reference [10].

C. INTERACTION CHAMBER LOSSES

Losses in the interaction chamber are difficult to evaluate. Different approaches should be used depending on the length of the chamber. The optimum length of the interaction chamber is defined as that length where the primary fluid has completed its energy transfer to the secondary fluid. Beyond that length the Darcy-Weisback equation should be used to account for friction losses. Up to the optimum length the Darcy-Weisback equation for friction losses in a pipe should be modified. The secondary fluid will enter the zone of interaction along the outer walls at extremely small velocities. As the energy exchange progresses, the secondary fluid velocity will increase to its maximum at the optimum mixing chamber length. Friction losses in the mixing chamber up to the optimum chamber length were evaluated by Reddy [Ref. 5] using the Darcy-Weisback equation and the average secondary velocity in the mixing chamber.

$$h_{mc} = f \frac{L_{mc}}{d_{mc}} \frac{\left(\frac{U}{2}\right)^2}{2g} \qquad (39)$$

where L_{mc} -mixing chamber length

d_{mc} -mixing chamber diameter

U -velocity of the combined primary and secondary fluids at the discharge of the mixing chamber.

Using the continuity equation (Eq. 3) and the area ratio α (Eq. 4).

$$\frac{U}{U_p} = \alpha + \frac{U_s}{U_p} (1-\alpha) \qquad (40)$$

substituting Equation (40) into Equation (39) produces the friction head loss for the mixing chamber.

$$h_{mc} = f\frac{L_{mc}}{d_{mn}} [\alpha + U_r(1-\alpha)]^2 \frac{U_p{}^2}{8g} \tag{41}$$

D. DIFFUSER LOSSES

Kinetic energy of the combined primary and secondary fluid is converted to pressure energy in the diffuser. Diffuser lossers fall into two categories: losses due to friction and losses due to diffusion of the fluid. Using the continuity, energy, and momentum equations, diffuser head losses were determined by Reddy [Ref. 5] to be

$$h_D = [\frac{f}{8} \cot\theta_D(1-\frac{1}{n_D{}^2}) + (\frac{n_D-1}{n_D+1})\sin 2\theta_D(1-\frac{1}{n_D{}^2})]\frac{U^2}{2g} \tag{42}$$

substituting Equation (40) into Equation (42)

$$h_D = [\frac{f}{8}\cot\theta_D(1-\frac{1}{n_D{}^2}) + (\frac{n_D-1}{n_D+1})\sin 2\theta_D(1-\frac{1}{n_D{}^2})][\alpha+U_r(1+\alpha)]^2\frac{U_p{}^2}{2g} \tag{43}$$

PRIMARY PRESSURE P_p (PSI)	PRIMARY VELOCITY U_p (ft/sec)	SECONDARY VELOCITY (ft/sec)	OUTLET VELOCITY U (ft/sec)	PRIMARY FLOW RATE Q_p (GPM)	SECONDARY FLOW RATE Q_s (GPM)	OUTLET FLOW RATE Q (GPM)	FLOW RATIO Q/Q_p	EFFICIENCY η_{jp}
13.0	34.7	1.49	2.03	5.3	14.59	19.90	3.74	1.88
15.0	38.6	1.74	2.34	5.9	17.00	22.90	3.88	2.12
15.7	39.9	1.80	2.42	6.1	17.60	23.70	3.88	2.12
18.0	42.5	1.84	2.50	6.5	18.00	24.50	3.76	1.92
18.3	43.8	1.82	2.50	6.7	17.80	24.50	3.66	1.73
19.5	45.1	1.87	2.57	6.9	18.30	25.20	3.65	1.72
19.7	47.4	1.92	2.65	7.3	18.75	26.00	3.59	1.62
Uncertainty								
$\pm$ .1	$\pm$ .1	$\pm$ .02	$\pm$ .02	$\pm$ .1	$\pm$ .05	$\pm$ .05	$\pm$ .02	$\pm$ .02

APPENDIX D: DATA

TABLE I: DATA FOR O DEGREE GLADE ANGLE NOZZLE

PRIMARY PRESSURE P_p (PSI)	PRIMARY VELOCITY U_p (ft/sec)	SECON-DARY VELOCITY U_s (ft/sec)	OUTLET VELOCITY U (ft/sec)	PRIMARY FLOW RATE Q_p (GPM)	SECONDARY FLOW RATE Q_s (GPM)	OUTLET FLOW RATE Q (GPM)	FLOW RATIO Q/Q_p	EFFICIEN-CY η_{jp}
15.6	32.0	1.49	2.03	5.3	1.50	2.03	3.77	1.92
18.4	34.7	1.65	2.19	5.3	1.65	2.19	4.06	2.46
21.0	37.3	1.84	2.42	5.7	1.84	2.42	4.12	2.66
23.5	20.0	2.18	2.81	6.1	2.18	2.81	4.51	3.48
24.5	42.5	2.45	3.12	6.5	2.45	3.12	4.70	3.98
Uncertainty								
$\pm$ .1	$\pm$ .1	$\pm$ .02	$\pm$ .02	$\pm$ .1	$\pm$ .05	$\pm$ .05	$\pm$ .02	$\pm$ .02

TABLE II: DATA FOR 20 DEGREE BLADE ANGLE NOZZLE

PRIMARY PRESSURE P_p (PSI)	PRIMARY VELOCITY U_p (ft/sec)	SECON-DARY VELOCITY U_s (ft/sec)	OUTLET VELOCITY U (ft/sec)	PRIMARY FLOW RATE Q_p (GPM)	SECONDARY FLOW RATE Q_s (GPM)	OUTLET FLOW RATE Q (GPM)	FLOW RATIO Q/Q_p	EFFICIEN-CY η_{jp}
12.4	28.3	1.36	1.80	4.3	13.30	17.64	4.08	2.50
13.3	29.2	1.50	1.96	4.5	14.70	19.16	4.30	2.98
17.6	32.0	1.84	2.34	4.9	18.04	22.94	4.68	3.93
22.0	36.9	2.38	2.96	5.7	23.35	29.00	5.13	5.31
25.0	40.0	2.73	3.35	6.1	26.69	32.79	5.38	6.17
26.0	41.2	2.86	3.50	6.3	28.00	34.30	5.44	6.43
Uncertainty								
$\pm$ .1	$\pm$ .1	$\pm$ .02	$\pm$ .02	$\pm$ .1	$\pm$ .05	$\pm$ .05	$\pm$ .02	$\pm$ .02

TABLE III: DATA FOR 35 DEGREE ANAGLE NOZZLE

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